



APPARATUS FOR FEEDING A GASEOUS AND/OR LIQUID MEDIUM TO A ROTATING
PRESSURIZED SYSTEM

CROSS REFERENCE TO RELATED APPLICATIONS

This application is the US national phase of PCT
5 application PCT/DE2003/004011, filed 4 December 2003, published 17
June 2004 as WO 2004/051130, and claiming the priority of German
patent application 10256537.6 itself filed 4 December 2002, whose
entire disclosures are herewith incorporated by reference.

FIELD OF THE INVENTION

10 The invention relates to an apparatus or device for
feeding a gaseous and/or liquid medium to a rotating pressurized
system, provided with a stationary seal-head housing with a
connection port for the medium to which a connection passage
extending into the seal-head housing is connected, as well as with
15 a tubular rotor connected with the pressurized system and also
extending into the sealhead housing, and furthermore with a stator
seal at an end of the connection passage and a rotor seal on an end
of the rotor, the stator seal and the rotor seal being coaxial to
the rotor and engaging each other.

20 An apparatus of this type serves for example as a seal
head or rotary feed coupling to transmit a fluid from stationary to
rotary machine elements and/or vice versa. As a result of
technical advances the requirements for rotary feedthroughs are
increasing regularly. In particular shop machines with spindle
25 rotation speeds have increased in the last decade from about 4000
to 6000 RPM to 20,000 to 50,000 RPM. In addition the pressures of

the media are steadily increasing. With minimal lubrication plus new applications, e.g. bowing off chips with compressed air when machining graphite, in order to get the medium to the tool it is necessary to meet extreme requirements in particular with respect to the seals.

Particularly critical with the increased demands on the service life of rotary couplings are leaks that can occur in the bearing or in the spindle and which blow out the seals, for example when they are pressurized but there is no actual flow.

In addition it cannot happen that as a result of standard use the service life of the known rotary feed couplings is reduced.

More specifically, the following factors have been found to have a particularly great influence on the pressure stress in a seal:

the pressure of the medium and the pressure on the seal surfaces;

the type of medium and how it lubricates;

the rotary speed and the diameter or frictional speed;

and

the throughflow speed and type of the medium and how it is cooled.

All these factors directly affect the generation of heat in the seal. Low temperatures are associated with low stress, high temperatures with high stress of the seal. Thus operating temperature is a significant factor affecting control of the pressure stress on the seal. It is said in control technology that temperature is the leading factor.

German 4,203,954 describes a rotary coupling where the rotor is axially positioned relative to the stator either manually or by means of a bimetallic element in dependence on temperature. The disadvantage of this solution is the substantial dependence of the adjustment movement and the adjustment action on the pressure and the setting force. In addition all the stresses on the seal vary independently of the respective loads.

OBJECT OF THE INVENTION

It is an object of the invention to provide an apparatus of the above-described type in the form of a rotary feedthrough coupling that has a long service life even when used in different application and that also is uniquely set up that independent of operating conditions, medium, and outside circumstances it works surely and is guaranteed to have a long service life. It is significant that the seal and bearing load be reduced so as to minimize any leakage.

SUMMARY OF THE INVENTION

This object is solved according to the invention in that near the stator and rotor seals there is a thermally influenced strain element that shifts the stator and rotor seals toward and/or away from each other in accordance with temperature.

The advantage of the instant invention is that the control reaction is very fast. On the one hand there is very fine control as a result of the longitudinal movement. On the other hand large forces resulting from the length change have a considerable influence on the control reaction, independently of the pressure of the medium. Advantageously in addition there is

the nearly unlimited and friction-free reproducibility of the thermally actuated length change.

In a preferred embodiment of the invention the strain element is formed as a sleeve that has an end turned toward the pressurized system and fixed on the seal-head housing and an
5 opposite end turned toward the stator seal and bearing against a coupling ring connected with the stator seal. Since the strain element completely surrounds the exact place where temperature is most critical, any temperature increase is immediately converted
10 into an adjustment action and a reduction of stress on the seals.

In order to further increase the sensitivity of the arrangement, the strain element has at least on its inner surface a heat-absorbing surface coating and/or surface increasing structure.

To ensure that the arrangement is as free of play as
15 possible it is further proposed that the coupling ring be urged by a spring against the strain element.

A further advantageous embodiment of the invention is that the coupling ring surrounds an axially shiftable seal support carrying the stator seal and a stator spring presses it and the
20 stator seal on the axially fixed rotor seal. In addition it is preferred that the seal support have an annular rim projecting radially toward the coupling ring.

In a particularly advantageous embodiment according to the invention, the coupling ring is made of a material with a low
25 coefficient of thermal expansion and the seal support of a material with a greater coefficient of thermal expansion and that at normal operating temperature there is a narrow annular gap between the

coupling ring and the seal support. in this manner the stator seal during normal operation is pressed by the stator spring against the rotor seal. Only when a certain temperature threshold is passed is there friction between the coupling ring and the seal support, so
5 that the strain element can be effective via the coupling ring and the seal support on the stator seal to relieve pressure at the seal faces without however creating a gap, which is not wanted so that particles cannot get in and create wear.

Furthermore it is preferred for heat-transfer reasons
10 that the annular rim of the seal support be conically tapered at its end toward the stator seal.

In order to prevent an unwanted actuation of the control mechanism, that is to prevent the pressure of the medium and the forces created by it from acting on the rotor and stator seals, the
15 outer diameter of the part of the seal support set in the seal-head housing is the same as the diameter of the rotor bore holding the rotor and the bore diameter of the rotor seal and the stator seal are the same. Thus these axially effective forces cancel each other out.

20 In order that the rotor seal does not move with the stator seal when it is being drawn back for control purposes, a spring presses the rotor seal axially against a rim.

It has further proven advantageous when flanking the bearing there are seal gaps each with a respective annular passage
25 to which a blocking medium, e.g. compressed air, is supplied and that have at least a partial connection to a leak fitting.

A further preferred embodiment of the invention is characterized in that between the strain element and the movably mounted seal support there is a rotary thermally actuatable coupling.

5 Preferably the coupling is formed by two relatively rotatable coupling disks of which at least one is formed as a wedge disk with an angularly extending wedge face. For relative rotation a spiral-shaped bimetallic element is secured at an end to one of the coupling disks.

10 At normal or room temperature the coupling disks are so positioned relative to each other that together they are of shortest possible axial length. A first coupling disk is connected to and rotatable by the strain element and the second coupling disk is engaged by the first. The bimetallic element that is connected
15 to the second coupling disk, has rotated this into a position in which the coupling is of minimal length and there no contact between the second coupling disk and the axially movable seal ring or its support.

If the temperature of the bimetallic element located
20 right at the seal joint increases, for example because of friction, the second coupling disk is pivoted on the wedge faces until there is actual contact with the axially movable seal or its support. When the flank angle is small there is a self-locking action even when high axial pressures are being transmitted and no further
25 rotation is possible. The longitudinal expansion of the strain element is effective directly to change the position or reduce the pressure in the seal.

According to a further embodiment of the invention a stress-relief ring is provided between the coupling ring and the seal support.

To this end the coupling ring is made of a material of low coefficient of thermal expansion and the stress-relief ring of a material of higher coefficient of thermal expansion and at normal operating temperature there is a small annular gap between the coupling ring and the stress-relief ring.

Further according to the invention a specially shaped annular rim of the stress-relief ring is continuously pressed endwise by springs against the seal support and heat is conducted by metal-to-metal contact between the seal support and the stress-relief ring.

The stress-relief ring presses lightly at all times against the seal support. The temperature-dependent adjustment of the stator and rotor seals is thus only effective when as a result of heat build up between the stress-relief ring and the coupling ring the radial gap is eliminated so that friction is created and the length of the strain sleeve changes. The difference between this and the previously described solution is that the surfaces effective for coupling and coupling are done are different. In the previously describe embodiment coupling and stress relief are on the contrary done by the adjacent surfaces of the coupling ring and seal support.

This modified embodiment has on the other hand a coupling between the coupling ring and the stress-relief ring and stress relief is effected between the seal support and the annular rim of

the stress-relief ring. This makes possible better movability of the seal support and thus better operational performance of the seal.

BRIEF DESCRIPTION OF THE DRAWING

5 In the following the invention is described more closely with reference to an embodiment shown in the drawing. Therein:

FIG. 1 is a section through the apparatus according to the invention;

10 FIG. 2 shows the apparatus of FIG. 1 built into the stator of a machine spindle;

FIG. 3 is a variant on the structure of FIG. 1, shown only partially;

FIG. 4 is a further embodiment of the FIG. 1 structure.

SPECIFIC DESCRIPTION

15 The apparatus shown in the drawing serves for conducting a gaseous and/or liquid medium to a rotating pressurized system. It basically comprises a stationary seal-head housing 1 with a connection port 2 for the medium opening into a connection passage 2.1 extending inside the seal-head housing 1. In addition it also
20 has inside the seal-head housing 1 and connected with the pressurized system a tubular rotor 3 that is journaled via two roller bearings 4 on the stationary seal-head housing 1.

At the end of the connection passage 2.1 there is a stator seal 5 while the end of the rotor 3 has a rotor seal 6, the
25 stator seal 5 and rotor seal 6 being coaxial with the rotor 3 and bearing on each other at end faces. The end faces can, as in the embodiment, be planar or have a conical or other shape.

Near the stator and rotor seals 5 and 6 there is a thermally adjustable strain element 7 that directly axially effects a temperature-dependent shifting of the relative axial positions of the stator and rotor seals 5 and 6. This shifting is done such that when the temperature around the seals 5 and 6 increases so as to increase wear, this is countered by a slight increasing of their relative spacing without however actually separating the two seals 5 and 6 from each other. This shifting is in practice in the micrometer range.

The strain element 7 is to this end formed as a sleeve that is fixed at its end turned toward the pressurized system on the seal-head housing 1. The end of the element 7 turned toward the stator seal 5 on the other hand is connected to a coupling ring 8 connected with the stator seal 5. With this type of connection the entire length of the strain element 7 is available for a thermally caused length change and thus the shifting of the seals 5 and 6.

The strain element 7 can also be provided at least on its inner surface with a coating to absorb heat radiation, so that it heats more quickly and provides a faster-acting control mechanism.

The coupling ring 8 is pressed without play by a spring 9 against the strain element 7. Furthermore the coupling ring 8 surrounds an axially shiftable seal support 10 in whose end the stator seal 5 is seated, and a stator spring 11 presses this stator seal 5 against the axially fixed rotor seal 6.

The seal support 10 is also formed with an annular rim 10.1 projecting radially toward the coupling ring 8, the coupling

ring 8 and the seal support 10 being constructed such that at regular operating temperature there is a small annular gap between them.

5 The coupling ring 8 is formed of a material having a low coefficient of thermal expansion while the seal support 10 is made of a material with a greater coefficient of thermal expansion. Thus in the event of heating, the seal support 10 will expand radially and decrease the annular gap until there is actual frictional coupling between the seal support 10 and the coupling
10 ring 8. Starting at this time the seal support 10 is entrained by the coupling ring 8 for joint longitudinal movement as it is shifted by the strain element 7 so that the stator seal is pulled slightly away from the rotor seal 6, while however no actual space is created between the two seals 5 and 6. This ensures that no
15 particles that could wear the seals 5 and 6 can get between the rotor and stator seals 5 and 6.

When subsequently the temperature drops, the strain element 7 shortens and the frictional coupling between the seal support 10 and the coupling ring 8 is lost. When heated up again,
20 the control mechanism responds again in the described manner.

In order also to prevent hydraulic forces that come from the heat-exchange medium from acting on the seals 5 and 6, the outer diameter of the part of the seal support 10 engaged in the seal-head housing 1 is the same as the diameter of the rotor bore
25 holding the rotor 3. In addition the bore diameters of the rotor seal 6 and the stator seal 5 are the same. This prevents that, for example as a result of pressure variations, addition forces

effective on the actual control mechanism have an influence on the seals 5 and 6.

Finally, flanking the two seals 4 there are seal gaps 12 with annular passages through which a blocking medium such as, for example, compressed air is supplied and which at least partially are connected to a leak connection.

FIG. 3 shows a further embodiment of the invention where between the strain element and the movable seal support 10 there is a rotationally effective and thermally actuated coupling. The coupling is formed by two rotatably relatively movable coupling disks 16 and 17 of which one is formed as a wedge disk with peripheral angled wedge faces 17.1. The second coupling disk 17 has axially projecting bumps not visible in the drawing that engage the wedge faces 17.1. The coupling disks 16 and 17 are rotated by a spiral-shaped bimetallic element 18 connected at an end of one of the coupling disks.

At rest or room temperature the coupling disks 16 and 17 are relatively so oriented that they are of minimal axial length. The one coupling disk 17 engages the strain element 7 and is rotatable. The bimetallic element 18 at room temperature holds the second coupling disk 17 in a position in which the coupling is of minimal length and there is no contact between the second coupling disk and the axially movable seal or the seal support 10.

When the temperature increases right at the bimetallic element 18 in the seal space, for example as a result of friction, the coupling disk 17 is rotated so far on the wedge faces until it contacts the axially shiftable seal or its seal support 10. As a

result of a limited wedge angle for self-locking action even with a great axial load there is no further rotational position change. The longitudinal shifting of the strain element 7 is now effective on positioning and reducing forces in the seal.

5 In the embodiment shown in FIG. 4 there is between the coupling ring and the seal support 10 a stress-relief ring 14. To this end the coupling ring 8 is made of a material with a low coefficient of thermal expansion and the stress-relief ring 14 of a material with a substantially larger coefficient of thermal
10 expansion, and at normal temperature there is between the coupling ring 8 and the stress-relief ring 14 a narrow annular gap.

 The stress-relief ring 14 bears at all times by means of a spring 15 on an end of the seal support 10 via a specially shaped annular rim 14.1, and as a result of metal-to-metal contact and
15 conduction the heat of the seal support 10 is transmitted to the stress-relief ring 14.

 The stress-relief ring 14 bears lightly at all times on the seal support 10. The temperature-dependent stress relief between the stator and rotor seals is only effective when as a
20 result of heating the radial gap is eliminated between the stress-relief ring 14 and the coupling ring 8, so that there is a frictional connection and also a length change of the strain sleeve 7. The difference from the other above-described solutions is that there is a separation between the surfaces effective for coupling
25 and relieving stress. In the already described solution the coupling and stress relief are done via the adjacent faces of the coupling ring 8 and seal support 10.

This modified embodiment however couples the coupling ring 8 and the stress relief ring 14. Stress relief is effected between the seal support 10 and the annular rim 14.1 of the stress-relief ring 14. This facilitates easier movement of the seal support 10 and a more reliable operation of the seal.